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Lubrication

A Technical Publication Devoted to
the Selection and Use of Lubricants

THIS ISSUE

The Relation of Theory
to Practice in Plain
Bearing Lubrication



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Effective BEARING LUBRICATION

In the attainment of effective bearing lubrication there are certain controlling factors which should be given the utmost consideration. They involve both constructional characteristics and operating conditions. They alone dictate the extent to which perfect lubrication can be approached. They include

- The load,
- The operating speed,
- The bearing clearance,
- The viscosity of the lubricant, and
- The point of oil application.

Theory of lubrication has proven that in practice a definite relationship exists between these factors.

The purpose of the following article has been to indicate this relationship and the manner in which the theory can be adapted to a more definite determination of the necessary viscosity to obtain the most effective lubrication.

In consequence, we commend this data for study. It is based on the research of the most eminent scientists in the field of lubrication. Their efforts have been untiring, and we deem it a privilege to be able to quote their opinions for the benefit of the readers of LUBRICATION.

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The Relation of Theory to Practice in Plain Bearing Lubrication

THÉORY of plain bearing lubrication has for years been the subject of discussion among engineers and research students who have interested themselves in oil film development and action.

In view of the fact that the theory has of necessity received the most attention, there is a prevailing impression, particularly among practical operators, that the explanation of the development of variable pressures within the oil film and their consequent carrying capacities is so deep and involved as to make it impossible for those of average mathematical training to understand. To be true, the average study of Bearing Lubrication appears complicated, but while it may have the appearance of an extremely extensive mathematical treatise, on the other hand, it has nevertheless brought out some decidedly practical points which we feel should be more clearly understood by the operating engineer.

Effective bearing lubrication is certainly an economic necessity, dictated by high standards of engineering efficiency and competition. Proper lubrication, after all, means longer life, less lubricant, lower maintenance and repair bills, lower unit costs and higher efficiency of conversion of available energy into useful work.

It is fortunate, for these reasons, that study has recently been devoted to the more ready adaptation of this theory to the practical solution of problems which are of vital interest to both designing and operating engineers.

In other words, these studies indicate that selection of an oil of estimated proper viscosity, based on past practical experience, can be am-

plified and checked in order to more nearly determine the most suitable viscosity oil to produce the lowest coefficient of friction for any speed, clearance and load combination when the bearing operates in the range of continuous fluid film lubrication; that is, when the rubbing surfaces are completely separated by a liquid film and the only frictional resistance to be overcome is the internal friction of the lubricant itself.

EMPIRICAL RULES OF FRICTION*

Bearing lubrication problems have existed ever since the development of our earliest machinery. As we proceeded to adopt more heavily loaded bearings operating at increasingly higher speeds, the designing engineers were confronted with the ever pertinent problem of how to most effectively provide for the maintenance of satisfactory lubrication.

Lubrication troubles of all sorts arose as time went on, and most of them found ready solution. The engineers who managed to solve the lubricating problems of their own particular piece of equipment at times were too eager to propound their particular remedies as conclusive evidences of some general law of lubrication. The result of this tendency to generalize was a mass of detached and inconsistent empirical rules which caused quite a bit of confusion.

The general trend of opinions seemed to be that total friction varied directly with the load for poorly lubricated bearings thereby approxi-

*"Report of Comm. on Friction" Proc. Inst. M. E. Nov. 1883; Kent 10th ed. pp. 1707.

mately following the laws of dry friction. But for perfect fluid film lubrication there was a marked difference of opinion. While some authors thought that the total friction varied as the square root of the total load, others

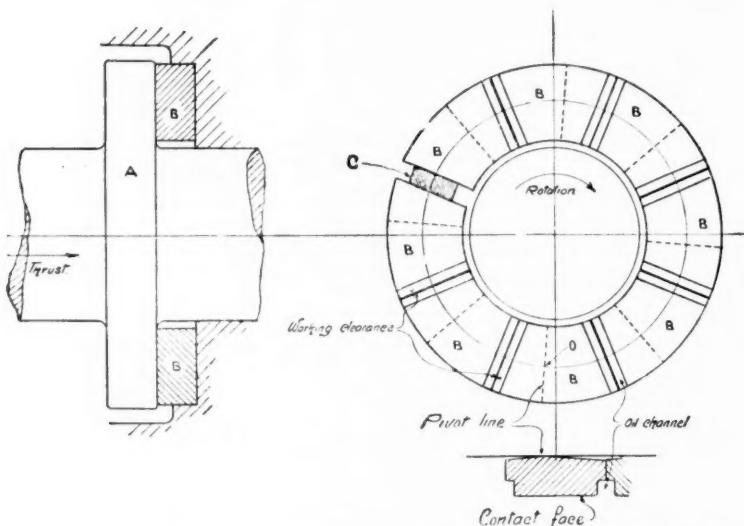


Fig. 1—Showing modern Michell Thrust Bearing and arrangement of thrust blocks.

saw no relation at all between total friction and load. These latter ones pointed out that coefficient of friction consequently had to vary inversely with load. Still other authors thought that the total friction varied directly with speed, while others thought it varied with the square root of speed.

These differences of opinions should cause the reader to wonder just how it was possible for competent and able engineers and experimenters to arrive at such widely different conclusions.

With the knowledge at our command now, we can point to the answer with relative ease. The early investigators just simply failed to recognize *all* the variables controlling the phenomenon of bearing lubrication, i.e., total frictional resistance, radius, length, clearance, eccentricity, load, speed, viscosity of lubricant, oiliness of lubricant, adhesive forces between oil and metal, amount of lubricant fed to the bearing, nature of the surfaces, materials of construction and temperature.

Osborne Reynolds' Hydrodynamic Theory of Bearing Lubrication*2

Professor Reynolds was interested in the experimental results of Beauchamp Tower, which showed the existence of considerable pressures in lubricating oil films. His mathematical treatment of the problem finally became the

*2 Reynolds generally is credited with the development of the theory. However, Stokes and Raleigh simultaneously arrived at the same results, while Petroff, too, worked along similar lines.

foundation of the hydrodynamic theory of lubrication.

Reynolds demonstrated mathematically, beyond a reasonable doubt, certain facts which before were only felt to be so, and other facts which perhaps were never thought of before. He showed, for instance, that the pressures within an oil film could not be uniform, and further showed how they varied within the oil film. He also demonstrated the distribution of oil pressures over the journal surface and predicted the exact location of the points of maximum and minimum pressures. He further showed that the wedge shape of the film was essential for the automatic generation of pressures within the film and the consequent load carrying capacity of such wedge shaped films.

On the other hand, Reynolds had to make certain assumptions in order to be able to establish these facts. These assumptions were as follows:

- (1) Uniform viscosity of lubricant throughout the film,
- (2) Freedom from oil leakage at the end,
- (3) No variation of pressures over the length of bearing and an abrupt drop to atmospheric pressures at the ends,
- (4) Smooth and geometrically perfect surfaces (freedom from distortion),
- (5) Absence of an external couple about an axis perpendicular to that of the axis of the journal, such as might be expected from belt pull,
- (6) Rate of oil feed low enough to eliminate thrust, due to change in momentum of the incoming oil stream, yet large enough to supply enough oil for the maintenance of a continuous film,
- (7) A continuous oil film separating bearing from journal,
- (8) An oil film thick enough to possess ordinary bulk properties so that no consideration of unknown "oiliness" factors was necessary, and
- (9) Perfect adhesion between oil and metal.

Conclusions to be Drawn from Reynolds' Work

Basing his mathematical reasoning on the above assumptions, Reynolds came to the following conclusions that:

LUBRICATION

- (1) The load which can be carried is directly proportional to the viscosity of the oil,
- (2) The load is also directly proportional to the journal speed,
- (3) The load which can be carried is inversely proportional to the square of the oil film thickness, and
- (4) That strictly parallel plane surfaces could not carry any load at all.

It has since been observed, however, that ordinary thrust blocks*³ with parallel surfaces do carry light loads quite successfully. Probably this is to be attributed to the fact that the surfaces are not absolutely parallel in the sense of the theory. In fact, there is always some rounding off at the edges of oil grooves, thus to a small extent approaching wedge shaped oil films at least in some parts of the bearing surfaces.

Influence of Theory on Design of Thrust Bearings

As a matter of fact, the design of modern thrust bearings has recognized the essential necessity for the wedge shaped films.

Oil films are caused to assume that shape by having the segments or shoes composing the stationary wearing surface so designed that under operating conditions they are very slightly inclined to the opposite and revolving wearing surface.

"The oil films between them are therefore wedge shaped, the thin end pointing in the direction of rotation of the plain bearing relative to the shoes. . . . The amount of shoe tilt, and therefore the shape of the oil film wedge varies automatically with the characteristics of the oil, with the thrust load, and with the revolutions per minute. This automatic adjustment results in minimum friction under all conditions"**⁴ of fluid film lubrication.

A thrust bearing enjoying great popularity in Europe is the Michell thrust bearing, which is designed along the same lines as the Kingsbury Thrust Bearing which is extensively used in this country. Fig. 1 shows the general construction of a Michell Thrust Bearing.

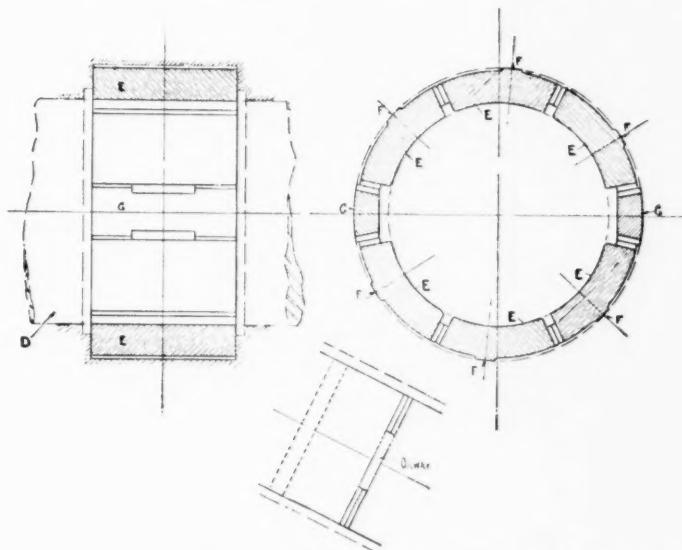
The principle of tilttable blocks has also been applied to the design of journal bearings. High safety at high speeds and unit pressures is claimed for this type of journal bearing by the manufacturers.

Fig. 2 shows the construction of a Michell Journal Bearing. "D" is the shaft, "E" is the

pivoted pad, "F" the pivot, and "G" are stays arranged to prevent rotation of pads.

Air As Lubricant

"The complete separation of the bearing



Courtesy of Michell Bearings, Ltd.

Fig. 2—Showing journal bearing of tiltable block design.

faces during operation prevents wear. The separation of the surfaces has been beautifully demonstrated by means of the small bearing illustrated by Fig. 3.

When lightly loaded it will run with air as the only lubricant. Air is much less viscous than oil but it separates the faces completely. This is proved by placing the air film in a battery circuit. . . . When the collar is revolving above a moderate speed, the current fails to pass across the air film to light a small lamp. As the collar slows down, the lamp will begin to flicker and finally as the collar comes to rest, it will burn steadily**⁴ Thus, by correct design "separation of surfaces is brought about regardless of the kind of lubricant employed,"**⁴ provided the speed is high enough for the load.

Cooling Maintains Load Carrying Capacity of Oil Films

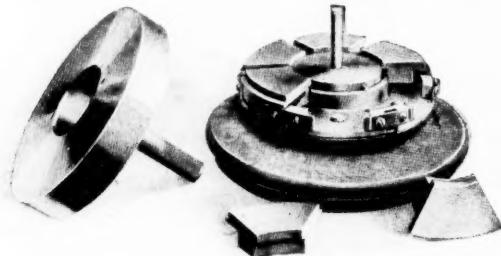
Reynolds' conclusion as to the proportionality between viscosity of the lubricant and the load that successfully may be carried has become one of the guiding principles underlying modern bearing design.

In the case of Kingsbury thrust bearings, we notice that**⁴ "Oil from the spaces between the shoes is continually supplying the film at the

^{**4} Kingsbury Thrust Bearing Catalog C, 1922, pp. 6 and 7.

*³ Eng'g., Vol. 100, pp. 154

thick ends. . . . Passage of oil through the film is accompanied by heating. The rate of heat generation in the bearing films is equivalent to the power required to overcome the frictional resistance to rotation. . . . It is desirable to cool



Courtesy of Kingsbury Machine Works.

Fig. 3—Experimental thrust bearing which operates successfully with air as lubricant.

the oil before passing it through the films again because the hotter the oil^{**4} the less its viscosity and consequently "the less is its load carrying capacity."

To maintain constant load carrying capacity a suitable means for cooling the oil should therefore be an essential part of any bearing design.

"One system of heat disposal, that has been widely used, provides for continuous circulation of the oil. The heated oil is withdrawn from the bearing housing, cooled and returned.^{**4}

"Another system, which is now frequently employed, provides for cooling the oil within the bearing housing by means of water passing through a coil of pipe. Such an arrangement is shown in section by Fig. 4."^{**4}

Of course, in some cases, the rate of heat generation may be small compared with the size of the bearing. In such cases where the heat carried off by the metal and given up to the air is as large as the heat generated, no additional means for oil cooling is required.

Extension of Reynolds' Theory

The practical illustrations mentioned above should convince even the most irreconcilable critics of theoretical thinking that this theory was shown to have possibilities. It pointed out some facts which readily found application in bearing design and lubrication. But there were still other facts, hidden away behind mysterious mathematical symbols, which threatened to remain secrets to many unless properly interpreted by subsequent investigators. And thus we now turn to the consideration of the work of those who extended Reynolds' studies.

^{**4} Kingsbury Thrust Bearing Catalog C, 1922, pp. 6 and 7.

Although these, without a doubt, really marked a milestone on the road to a fuller understanding of bearing lubrication, his work was far too theoretical to be understood by the designing and operating engineer. Much simplification of the work was needed before all those who were vitally concerned with lubrication could understand it.

Indeed the effort of the later authorities was directed along these lines. Sommerfeld^{*6} and Harrison^{*7} independently simplified and extended the theory.

Important Conclusions Drawn from Harrison's and Sommerfeld's Work

They pointed out that a journal running in a full bearing undergoes a displacement in a direction at right angles to the line of load, and that the point of nearest approach between journal and bearing is on a diameter at right angles to the line of load, as illustrated in Fig. 5.

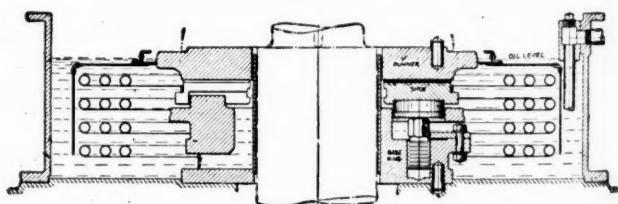
They further showed that the location of the point of maximum and minimum pressures was symmetrical with respect to the point of nearest approach. (See Fig. 6.)

They also showed the manner in which film thickness and pressures within the oil film varied over the journal surface. This is brought out in Fig. 7.

Plotting the developed pressures in an oil film around a circle representing the journal surface, the curve in Fig. 8 was obtained.

Fig. 8 is therefore a graphical presentation of the distribution of the calculated values of pressure over the journal surface. The vertical upward resultant of all these pressures is equal to the bearing load and thus is the equilibrant of the system.

It may be well for us to pause in the development of the theory and examine Fig. 8 more



Courtesy of Kingsbury Machine Works.

carefully. For instance, if the direction of the load against the bearing is known, (downward in Fig. 8), then the curve tells us that the lowest pressures in the oil film exist somewhere near

^{*6} Ztsch., f. Math. Phys., Vol. 50, pp. 97-155, 1904.

^{*7} Trans. Phil. Soc., Cambridge, 1913.

L U B R I C A T I O N

the center of the opposite bearing half, and it is there that we should logically locate the oil supply line when the film completely surrounds the journal.

If the oil supply line were erroneously located, say near "C", then the bearing would refuse to take up any further oil once the generated pressure at "C" exceeded the supply pressure. This at once limits the amount of lubricant that can be fed into the bearing, which may result in faulty lubrication, increased bearing temperatures and excessive wear.

The direction of load against the bearing is not known in a good many cases of machinery. In such cases it is impossible to predict the exact location of the minimum pressure and the best location for the oil feed line. The old method of trial and error would have to be resorted to here in order to finally obtain the position for the oil supply line, which would result in lowest bearing temperatures, highest carrying capacity and possibly a substitution of lower viscosity oil.

Oil Grooving

From the curve in Fig. 8 we have further to infer that anything that in any manner interferes with the attainment of full pressures within the film will cause the bearing to run inefficiently; that is, lower its carrying capacity. It is exactly for this reason that oil grooving in the high pressure zone would appear to be poor practice. Oil grooves interfere with the development of the pressure zone in the oil film since they permit the oil to escape too readily.

Oil grooves in the low pressure zone interfere very little, if at all, with the generation of

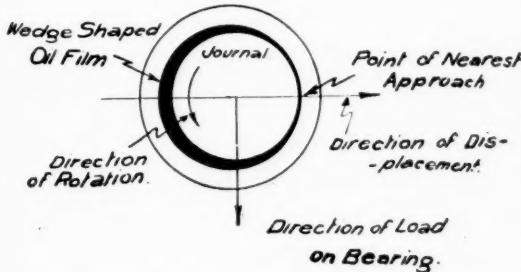


Fig. 5—Diagram showing wedge-shaped oil film, point of nearest approach, direction of displacement and line of load on bearing.

the necessary pressures within the oil film, and, on the other hand, they permit ready introduction of the lubricant into the bearing. Thus the theory appears to explain the successful practice of placing of oil grooves in the low pressure zone.

Graphical Presentation of the Theory of Bearing Lubrication

In spite of the efforts to simplify and to generalize the mathematics of the theory, it still was too formidable to encourage general

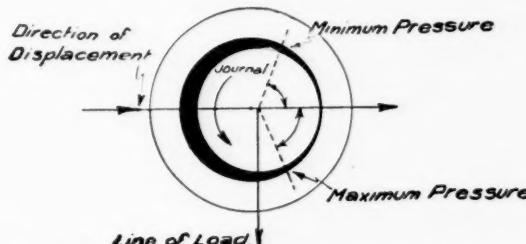


Fig. 6—Diagram showing location of maximum and minimum pressures in a full bearing.

study. This prompted Howarth to transform the mathematical analysis into a graphical analysis of almost handbook simplicity in order to bring the results of the theory before the eyes of operating and designing engineers, so to speak.

This graphical analysis, as published in the various journals, is already condensed, and for that reason, perhaps, difficult to further abstract. A complete presentation of this work thus appears to us beyond the scope of this article.

We will therefore confine this discussion to consideration of certain typical curves, offered by Mr. Howarth,*⁹ and the conclusions that can be inferred from them.

Plain Full Bearings

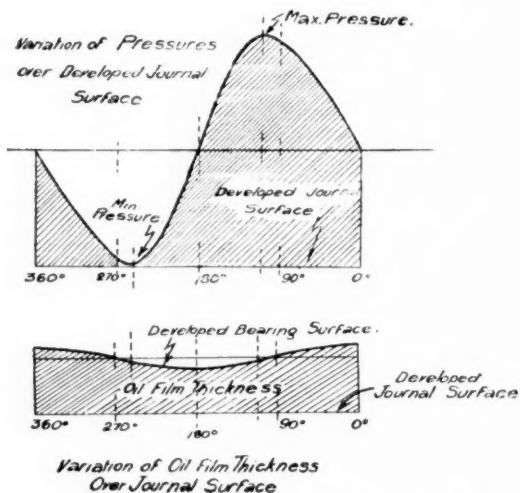
Reference to Fig. 9 shows that along the vertical axis, values of the ratio of load/speed are laid off. There are good reasons for working with the ratio of these two quantities, load and speed, as a single variable, in preference to working with load and speed individually.

The curve, to be useful, must be simple, and yet must completely tell the story. It would be possible to construct a curve plotting load against eccentricity factor, provided we kept the speed constant. Thus the curve could be constructed, for instance, showing how the eccentricity factor of the journal varied with load for a given viscosity clearance ratio and a speed of, say, 1000 R.P.M. Such a curve, however, would be entirely useless in case we should desire some information for the same journal when running at some other speed. A separate curve, other things being equal, would have to be constructed for each different speed.

If a series of such curves were plotted it would be found that no matter how large or how small the load and the speed were, so long

*⁹ "A Graphical Analysis of Journal Bearing Lubrication, presented at Annual Meetings of A. S. M. E. 1923, 1924, 1925.

of course, as they were within operating limits, the ratio of the two would always be the same for any particular eccentricity factor at which the journal may be running in the bearing. And since this ratio of load to speed is con-



Courtesy of American Society of Mechanical Engineers.
Fig. 7—Showing theoretical distribution of oil film and pressures in a full bearing.

stant for any eccentricity factor while clearance ratio and viscosity are constant, it is preferable to plot this ratio instead of speed or load individually, against the eccentricity factor.

Thus, by working with the load/speed ratio, a single curve will supply information for an infinite number of load and speed combinations. The load/speed curve may conveniently be called the loading curve¹⁰ since it enables us to determine the carrying capacity of any full bearing at any speed.

From Fig. 9 we can readily see that for the same eccentricity and clearance ratio, load/speed ratio increases as the viscosity of the oil increases. We can further see that increasing speed (keeping clearance ratio and viscosity constant) will cause the value of this ratio load/speed to decrease and consequently cause the journal to run more nearly concentric. Similarly, we can observe that a reduction in speed would result in an increase of eccentricity.

The curve in Fig. 10 indicates the manner in which the coefficient of journal friction varies with eccentricity. Thus, we can obtain from Fig. 9 the eccentricity at which a journal will run once we know the load, the speed, clearance ratio, and the viscosity. And from Fig. 10 we can then exactly determine what the coefficient of friction will be for these particular operating conditions.

By a method of trial and error we could thus determine the coefficient of friction obtained by

¹⁰ J. Ind. Eng. Chem., 1926.

the use of various viscosity oils and then pick out the one giving the lowest coefficient of friction; that is, the most suitable oil for the particular purpose.

The graphical work of Mr. Howarth also indicates that for any combination of coefficient of friction, clearance ratio and eccentricity, the ratio of load/speed will decrease when the clearance ratio is increased. His work further showed a method whereby the best clearance ratio, giving lowest coefficient of friction, can be graphically determined for any combination of speed, viscosity and load. In general, the curve between clearance ratio and coefficient of friction is of the shape shown in Fig. 11.

Practical Problems

A few practical problems relating to full bearings will now be presented and solved by means of the exact forms of the general curves shown above.

Problem 1

What is the coefficient of journal friction for a full turbine bearing when the diametric running clearance ratio is .002, the speed is 600 r.p.m., the main pressure is 120 pounds per square inch and the oil has a viscosity of 100 Seconds Saybolt Universal at the operating temperature?

$$\frac{\text{bearing pressure}}{\text{r.p.m.}} = 120/600 = .20$$

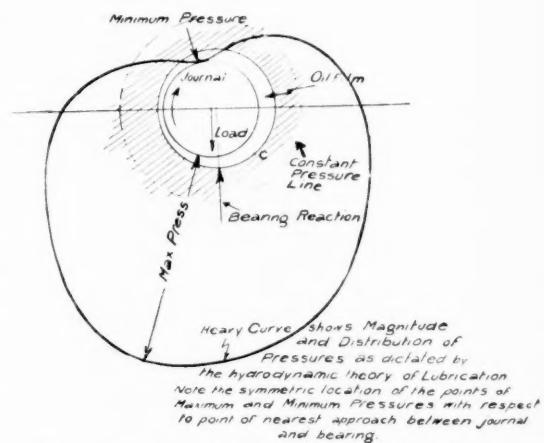


Fig. 8—Showing theoretical distribution of pressures around journal in a full bearing.

According to the above curve, this journal will run at an eccentricity of 37%. The coefficient of friction corresponding to this eccentricity is .0023.

Problem 2

What should be the viscosity of the lubricant to be used in a turbine bearing operating under the following conditions?

L U B R I C A T I O N

Load 63 pounds per square inch
Speed 600 r.p.m.

Diametric Running
Clearance Ratio .004

$$\frac{\text{load}}{\text{speed}} = 63/600 = .15$$

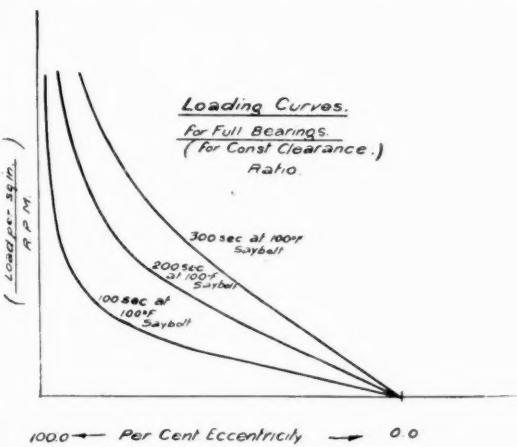
Viscosity in Seconds	Per cent Saybolt Universal at Operating Temperature	Coefficient of Friction
100	83%	.0038
150	65%	.0035
200	58%	.0039
250	49%	.0043

This table indicates that for the lowest coefficient of friction under the given operating conditions, an oil has to be used which has a viscosity of 150 Seconds Saybolt Universal at the operating temperature. This best viscosity, of course, is obtainable by proper selection of grade of lubricant or proper control of bearing temperature wherever this is possible.

Problem 3

What clearance ratio will give best performance, that is, lowest operating temperature and coefficient of friction for a 3" journal revolving in a full bearing at 1000 r.p.m. and under a load of 100 pounds per square inch when the lubricant to be used is to be of a vis-

cosity of 150 Seconds at the operating temperature?



Courtesy of American Society of Mechanical Engineers.
Fig. 9—Loading curves for full bearings.

cosity of 150 Seconds at the operating temperature?

$$\text{load} = 100/1000 = .10$$

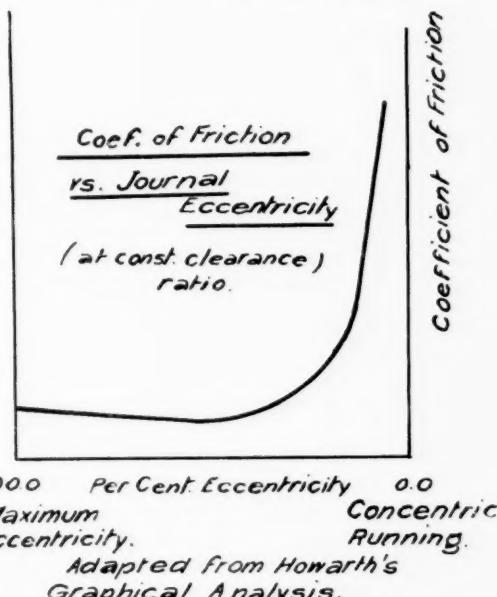
Clearance Ratio	Per cent Clearance	Closest Eccentricity Approach	Coefficient of Friction
.001*	.003"		.010*
.002	.006"	13	.0057
.003	.009"	27	.0040
.004	.012"	47	.0042
.005	.015"	67	.0049
.006	.018"	82	.0056

*Inches per inch of Journal Diameter.

Obviously, the most desirable diametric clearance ratio in this case is .003.

Problem 4

How closely will a 3" journal approach a full



Courtesy of American Society of Mechanical Engineers.
Fig. 10—Curve showing relation between coefficient of friction and eccentricity for a full bearing.

turbine bearing when the speed is 600 r.p.m., the load is 120 pounds per inch, the diametric running clearance ratio is .002 and the viscosity of the lubricant is 150 Seconds at the operating temperature?

$$\frac{\text{load}}{\text{speed}} = 120/600 = .20$$

Answers:

Per cent Eccentricity	24%
Running Clearance	.006 × .002 = .006"
Eccentricity	.006 × .24 = .00144"
Nearest Approach	.006 - .00144 = .00456"

Problem 5

Same as Problem 4, but a lubricant of 100 Seconds Saybolt Universal at the operating temperature is used here.

$$\frac{\text{load}}{\text{speed}} = 120/600 = .20$$

Answers:

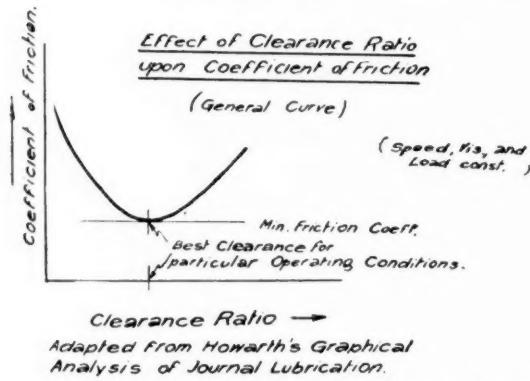
Per cent Eccentricity	38%
Running Clearance	.006"
Eccentricity	.006 × .38 = .00228"
Nearest Approach	.006 - .00228 = .00372"

These last two problems illustrate in a very interesting manner the fact that thinner oils permit closer approach between the wearing surfaces than thicker oils do.

Plain Partial Bearings

The same author also published a series of curves for this type of bearing.

From the shape of the curve in Fig. 12 we have to infer that the larger the bearing angle



Courtesy of American Society of Mechanical Engineers.
Fig. 11—Curve showing variation of coefficient of friction with clearance ratio.

the larger will also be the value of the ratio of load/speed permissible for any given viscosity and clearance ratio, if the coefficient of friction is to remain constant.

The exact forms of the curves shown in Figs. 12 and 13 permit us to solve similar problems, as indicated for full bearings.

Problem 6

What would be the coefficient of friction for a centrally loaded 120 degree partial turbine bearing operating under the following conditions?

Speed	600 r.p.m.
Load	120 pounds per square inch
Diametric Running Clearance Ratio	.002
Viscosity of the Lubricant	150 Seconds Saybolt Universal at operating temperature
load speed	$= 120/600 = .20$

According to the curves, the per cent eccentricity would be 64% and the corresponding coefficient of friction would be .0020.

All the other problems solved above could, in a similar fashion, be solved for centrally loaded partial bearings, as well as for unsymmetrically loaded partial bearings.

Main Conclusions to be Drawn from the Graphical Analysis

The last few pages show conclusions that may be drawn from this graphical analysis referred to above. It should be recalled, however, that these curves were built up from the equations developed by Reynolds, Harrison and Sommerfeld and thus, of course, are subject to

the same limiting assumptions as the original mathematical analysis itself.

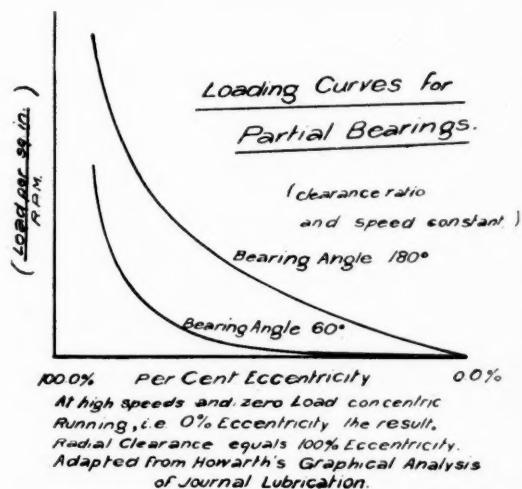
By means of this work, however, we can form a clearer picture of the relationship between such variables as load, speed, viscosity, etc.

JOURNAL RUNNING POSITIONS

Mr. Howarth has very recently extended his graphical analysis to the study of journal running positions. The question was whether or not a journal center could ever rise above the bearing center. This phase of the problem was subject of much experimental work by other workers as well.

In the discussion that followed the first part of Mr. Howarth's paper on "Graphical Study of Journal Lubrication" (presented before the A.S.M.E. in 1923), Professor G. H. Marx of Stanford University mentioned experimental evidence on "complete cylindrical bearings" which showed that the point of nearest approach did not lie on a line at right angles to the line of pressure, and that "in every case the center of the journal was lifted above the center of the bearing while the load was directed vertically downward."*11 Professor Marx did not try to explain this phenomenon but merely reported it "as an observed fact which seems to make untenable the fundamental assumptions of Harrison."*11

Mr. Howarth, however, was not ready to accept the results as conclusive evidence against



Courtesy of American Society of Mechanical Engineers.
Fig. 12—Loading curve for centrally loaded partial bearings 60 degrees and 180 degrees in length, respectively.

the classical theories for he thought that "the bearing to which he (Professor Marx) refers actually had two large oil ring openings in the upper half, which could reasonably be expected to prevent the realization of the ideal pressure

*11 Trans. A. S. M. E. 1923, pp. 437.

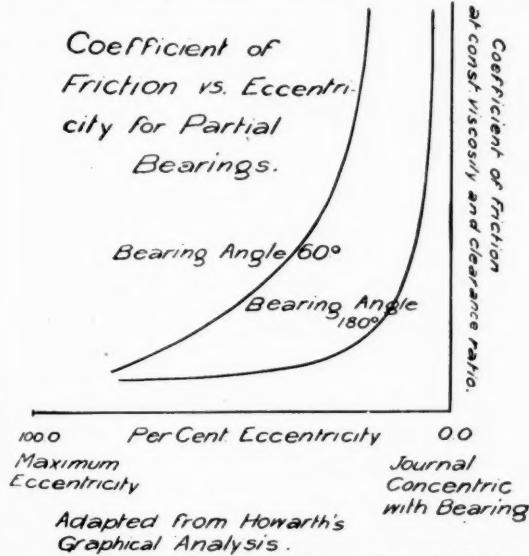
LUBRICATION

conditions. The unit pressures employed were very small and the method employed for charting the relative axis displacements would not show up such inconsistencies in the results as might actually exist. Consequently, those theses should not be accepted as discrediting the theories developed by Reynolds and others.*¹¹

However, some time later, Howarth published the results of his investigation of this phase of the problem of journal lubrication.*¹² He found that while according to theoretical reasoning the journal center could never rise above the center of a full bearing, there were certain conditions of speed, viscosity and load where the journal center could rise above the center of partial bearings provided the latter were of a certain design. He found that in the case of a symmetrically loaded 120 degree bearing the journal center never rose above the bearing center; in unsymmetrically loaded partial bearings, however, this may occur.

Journal Running Positions and Bearing Cap Design

"Bearing cap construction and position both



Courtesy of American Society of Mechanical Engineers.

play an important part in the operation of the journal. . . . The cap may exert pressures upon the journal that will partly determine its running position. Three distinct types of caps will be considered,"*¹² viz.:

*¹¹ Trans. A. S. M. E. 1923, pp. 437.

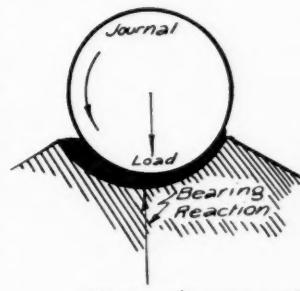
Cap "A"

The surface of which is concentric with the bearing.

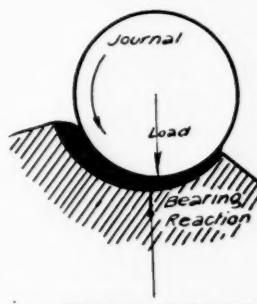
Cap "B"

The surface of which is concentric with

Central Partial 120° Bearing



Offset Partial 120° Bearing.



These diagrams show in an exaggerated manner the wedge-shaped oil films which by virtue of pressures generated within them can carry loads.

Fig. 14—Diagram showing difference between central and offset partial bearings.

running position of the bearing.

Cap "C"

The surface of which is offset and not concentric with either the bearing or the journal.

Howarth examined these three types of caps with reference to 120 degree partial bearings, in which the leading part of the bearing angle was 6/10 of the total bearing angle, and whose running clearance ratio was .001. He further assumed that the caps were also 120 degrees long, and that the oil channels at the sides were 60 degrees long each. His conclusions were as follows:

Cap "A"

A cap placed in this position will generate pressures in the cap film. "Their net effect would be to draw the journal toward the cap provided the naturally negative pressures in the film are not impossibly high. Presumably, the negative pressures could not exceed 14.7 pounds per square inch. If not, the film under the cap would break when that negative value was reached, and then the positive pressures under the cap might dominate, thereby opposing closer approach of the journal. Obviously, this needs experimental verification."*¹²

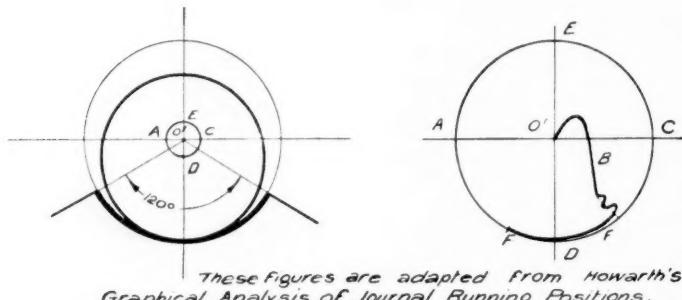
Cap "B"

When the bearing cap is concentric with the running position of the journal no pressures can be set up in the cap film and consequently the presence of the cap can exert no influence upon the ultimate position of the journal in the bearing.

*¹² "Journal Running Positions" Applied Mechanics Section Trans. A. S. M. E. 1929.

Cap "C"

"Obviously, if it is desirable to reduce the natural lift of the journal, a cap could be so placed as to generate only positive pressures that would oppose the normal lift."*12 This can



These Figures are adapted from Howarth's Graphical Analysis of Journal Running Positions. Heavy curve B in right-hand Figure shows path of Journal center.

Courtesy of American Society of Mechanical Engineers.

Fig. 15—Curve "B" in right-hand figure shows the general nature of path along which journal center may move.

be accomplished by shifting the cap sideways. The exact amount of shift required to prevent the journal from rising above the journal center can be calculated according to the graphical method indicated in Howarth's paper.

Path Described by Journal Center

Let these two eccentric circles in Fig. 15 (left) represent the full bearing and the journal at rest. O' is the center of the bearing, D is the center of the journal, and let Fig. 15 (right) be a larger scale picture of the clearance circle A, E, C, D. The 120 degree sector of the circle then would cover the case of the 120 degree partial bearing.

When the journal speeds up from rest in such a 120 degree partial bearing "its center presumably starts at some point "D", directly below the bearing center O'. Before the film is formed the journal will roll around inside the bearing until its center reaches some point "F" (left) determined by the coefficient of starting friction. Slip will then take place, and if the acceleration is rapid, a thin film will form at once. The journal center will, at the same time, slip over to some point near "F" (right).**12

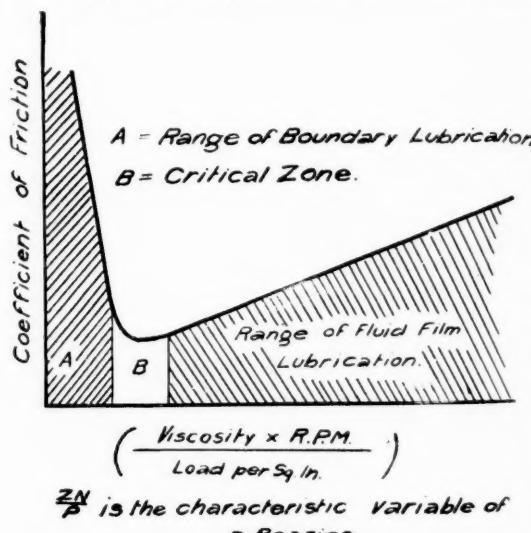
"When this slip takes place the coefficient of friction will fall to a very low value, and there may be some oscillation of the journal across its normal position for the speed at which it is rotating. As the journal gathers up speed its oscillation should cease, and the journal center should rise along a path-like curve "B" toward the bearing center O' as the oil film thickens.

Our analysis indicates that at infinite speed the journal and bearing centers will coincide. This statement appears to be true, if the effect of friction upon the direction of the resultant pressure be neglected."**12

THE RELATION BETWEEN LOAD, SPEED, VISCOSITY AND THE COEFFICIENT OF FRICTION

From a purely academic point of view, Mr. M. D. Hersey*13 made an investigation of the variables that control the phenomenon of bearing lubrication. He came to the general conclusion that for geometrically similar bearings the coefficient of friction could only depend directly upon a certain factor, $\frac{ZN}{P}$, where Z denotes viscosity, N the revolutions per minute, and P the bearing pressure. This meant, then, that the coefficient of friction would undergo the same change if the viscosity were doubled or the speed were doubled, or possibly the bearing pressure were halved.

Actual determination made on the coefficients of friction for various operating conditions give curves of the general type represented in Fig. 16. In reality, however, as pressures are increased, the coefficient of friction drops to a very definite minimum value,



$\frac{ZN}{P}$ is the characteristic variable of a bearing.

Fig. 16—This curve shows relation between coefficient of friction and viscosity, r.p.m. and bearing pressure.

beyond which it does not drop and from where on further increases in the pressure result in a sudden rise in the value of the coefficient of friction.

*12 "Journal Running Positions" Applied Mechanics Section Trans. A. S. M. E. 1929.

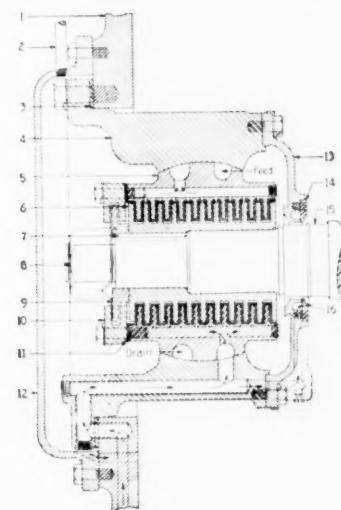
**13 Trans. A. S. M. E. 1915, pp. 167.

LUBRICATION

Thus, the straight line relationship between the coefficient of friction and $\frac{ZN}{P}$ ceases to exist at a very definite value of speed, load and viscosity. This point is now thought to be the beginning of a transition from perfect film lubrication to the range of discontinuous film type of lubrication. In other words, film rupture is thought to begin there. The shape of the curve indicates that the rupture is not sudden, but rather a gradual process. That portion of the curve where we have rapid changing of slope is generally referred to as the *critical zone*, while the right half of the curve is referred to as the zone of *fluid film lubrication*; the left portion is referred to as the range of discontinuous film type of lubrication, or *boundary lubrication*.

In the right half of the curve the laws of fluid friction apply, while in the left hand portion the laws of dry solid friction are approximated. Such curves, as shown in Fig. 16, have been experimentally determined for a large number of plain bearings, and it has been found that the nature of the metals, the finish of the surfaces, the length and diameter, the extent of the oil film, the degree of wear, and the clearance ratios seem to affect the slope of the right hand portion of the curve quite appreciably. It also has been observed that grooving tends to cause the right half of the curve to become convex upward,*14 while larger clearance ratios in general tended to lower its slope. In a paper by R. E. Wilson and D. P. Barnard

tion, the effect of oiliness upon this curve. It seems to be the modern belief that oiliness is of no consequence in the fluid range. This belief is in entire agreement with the results of Herschel*15 and others, according to which even



Courtesy of General Electric Co.

Fig. 18—Sectional drawing of self-aligning modern type thrust bearing for steam turbine.

poor lubricants, such as glycerin, and even non-lubricants, such as sugar water, gave the same results as the best lubricants in the region of fluid film lubrication. However, it seems that oiliness affects the value of the coefficient of friction in the critical zone by lowering this latter somewhat. That would indicate that somewhat larger reduction of speed and viscosity or correspondingly larger increases in load can be made before rupture of the oil film will take place.

Thus it would appear that more "oily" oils would increase the factor of safety of a bearing. It was also observed that by operating at a given speed, load and viscosity, smaller clearances seemed to facilitate the formation and maintenance of a continuous film since they produced critical zones at smaller values of $\frac{ZN}{P}$. This, in other words, then would mean that higher loads may be expected to be carried at lower viscosity and speed.

DISCREPANCIES BETWEEN THEORY AND PRACTICE

We have attempted in the preceding pages to develop the relationship between the different variables as predicted by the theory for an ideal bearing. As this theory was investigated and more nearly understood by a larger number

*14 "Bearing for High Speed Traction and Transmission" O. Lasche, 1903.

*15 J. S. A. E., Jan. 1922, pp. 31.



Schematic Presentation of the random distribution of oil molecules at some distance from metal faces.



Schematic Presentation of Orientated Oil Molecules immediately adjacent to Metal faces.

Fig. 17—Exaggerated presentation of film consisting of orientated oil molecules.

it was suggested that the exact shape of the curve for any bearing could be considered its characteristic.

Effect of Oiliness

It might be well to mention, in this connec-

of practical men, certain discrepancies between theory and practice were brought out.

Thus, as we have previously mentioned, it is almost impossible to predict the exact position of minimum pressure in the oil film.*¹⁶

It was also found that actually the pressure distribution throughout the film was by no means symmetrical about the point of nearest approach and that the maximum positive variable pressure was much larger than the corresponding negative pressure.*¹⁶ For short bearings, the displacement is not at right angles to the load.

It was also found that certain metals were more suitable for bearing construction than others.

The weakness of theory lay right in its simplifying assumptions. Thus, for instance, it had to be admitted that freedom of end leakage in reality was not found but merely approached in the case of very long bearings. Efforts to mathematically take into account this end leakage are on record*¹⁷ but they are of a rather involved mathematical nature and much beyond the scope of this article.

It might be sufficient to state that these mathematical efforts, attempting to take into account end leakage were limited to inclined plane surfaces sliding past another. Michell*¹⁷ found the ratio of "a" to "b" controlled the amount of friction developed where

"a" = length of plate in direction of motion

"b" = the width of the plate

From an analysis of Michell's work, Howarth showed that the least friction occurred when the length of the plate was 6/10 of its width.

CHEMISTRY OF LUBRICATION

There are a great many researches and technical papers on record which seem to cast new light on our knowledge of lubrication. These theories delve right into the chemical structure of oil molecules and metal molecules. Thorough understanding of these theories would require such an extensive chemical background on the part of the student as to be beyond the understanding of the non-chemical layman.

Some of the essential points brought out by these researches were that apparently the wearing metal surfaces actually are the seats of fairly powerful forces. These forces, emanating from the metal surfaces, exert a very definite effect on the oil molecules, in tending to rearrange them in the layers closest to the metals. Thus, while at greater distances from the metal faces, the distribution of the molecules is a random one, as shown in Fig. 17, there is a marked orientation of the molecules

*¹⁶ D. P. Barnard Trans. A. S. M. E., 1923, pp. 434.

*¹⁷ Michell, Ztsch. F., Math. u. Physik., Vol. 52, 1905.

in the layers immediately adjacent to the metal surfaces, in some manner, as indicated. The actual sliding motion is then pictured as taking place along plane X and X¹ formed by the "ends" of the orientated molecule. It has been shown that the friction developed along such a plane is of minimum value.

It is evident then that layers of oil at different distances from the metal surfaces differ in the degree of orientation of the molecules composing them. This might readily be expected to result in differences in properties throughout the full thickness of the oil film. This has caused some investigators to question the justification for the assumption No. 1 on Page 62, which presupposed the uniformity of properties throughout the oil film.*¹⁸

Now the effectiveness with which the orientating forces emanating from the metal surfaces will affect the oil molecules, of course, also depends on the chemical structure of the latter. It has been shown that properly refined lubricating oil contains certain constituents which are readily capable of orientation, and of thus producing a minimum coefficient of friction, while over-refined mineral oils are almost devoid of such constituents. This suggests the *essential need of proper refining of lubricating oils*.

The fact that different molecules are orientated to varying extents has caused other workers to question the justification of assumptions No. 8 and No. 9 on Page 62, where it was presupposed that all oils and all metals would behave alike.

CONCLUSION

We have, in the preceding pages, tried to sketch a picture of the theory in such a way as to make it readily understandable by practical operating men not already familiar with it.

The theory has pointed out that oil films in bearings are wedge-shaped, and that by virtue of this shape, pressures are developed within them which then permit them to carry loads successfully. The graphical form of the theory permits ready calculation of correct viscosity or clearance ratio for any definite operating condition.

It was further seen that although the theory was successful in explaining many phenomena, there still were discrepancies between theory and fact, which so far have defied all rigid mathematical analysis.

The essential point to keep in mind, however, is that, in spite of the different discrepancies, this theory has developed definite facts which have become the guiding ideals in modern bearing construction and bearing lubrication.

*¹⁸ Petroleum Ztsch., February, 1929.